Copy
RM E9L05

[redacted]

NACA RM E9L05

UNCLASSIFIED

NACA

RESEARCH MEMORANDUM

EFFECTS OF OBSTRUCTIONS IN COMPRESSOR INLET ON BLADE VIBRATION IN 10-STAGE AXIAL-FLOW COMPRESSOR

By André J. Meyer, Jr., Howard F. Calvert
and C. Robert Morse

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON
February 13, 1950

[redacted]

UNCLASSIFIED

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

EFFECTS OF OBSTRUCTIONS IN COMPRESSOR INLET ON BLADE VIBRATION IN 10-STAGE AXIAL-FLOW COMPRESSOR

By André J. Meyer, Jr., Howard F. Calvert
and C. Robert Morse

SUMMARY

A blade-vibration survey was conducted during normal operation of an early production 10-stage axial-flow compressor powered as part of a complete jet engine. Strain-gage records were obtained from all stages during acceleration, deceleration, and constant-speed runs. Curves are presented showing the effect of inlet disturbances caused by blocks at the compressor inlet on blade vibrations, the effects of inlet-disturbance location, and a comparison of maximum allowable stress to vibratory stresses measured. Values of aerodynamic damping were also estimated.

It was found that inlet disturbances affected blade vibrations throughout the compressor, that obstructions could be so located as to reduce some vibrations, and that aerodynamic damping accounted for about four-fifths of total dynamic blade damping.

INTRODUCTION

Blade failures in axial-flow compressors experienced in a number of different aircraft engines can undoubtedly be attributed either to excessive centrifugal or bending stresses caused by normal operation or to overspeed; many of the failures, however, were apparently caused by vibrations. The cause of these vibrations is often unknown.

The NACA Lewis laboratory therefore undertook to measure compressor-blade vibrations and to determine the effect of blocking the passages between inlet guide vanes on vibration characteristics. This blocking simulated ice formation or other foreign material lodging at the compressor inlet.

[redacted]

Data showing the effect of blocked inlet passages on blade vibrations and the aerodynamic damping as determined by this investigation are presented herein.

APPARATUS AND PROCEDURE

The jet-propulsion engine described in reference 1 was used for this investigation. The instrumentation used to obtain the data presented was of the type described in references 1 and 2 except for the introduction of a pure silk covering, which was cemented over the gages and lead wires to serve as a reinforcing material.

Vibrations at various critical speeds of the blades of each stage were measured throughout the complete operating range of the engine with and without fluid-flow blocks in the inlet section in order to evaluate the vibration characteristics possibly present under severe icing conditions or under conditions created by foreign matter lodging in the inlet of a jet engine. The blocks were sheet-metal plates with clips for attachment to two adjacent inlet guide vanes (fig. 1). Each installed inlet block prevented air flow in one of the 56 passages. Combinations of one, two, three, and four blocks were used. When more than one block was used, they were, as nearly as possible, installed with equal circumferential spacing. The inlet blocks were haphazardly oriented with respect to front bearing supports and other engine parts except in one run in which two diametrically placed blocks were successively moved to every other inlet passage around the annulus.

Resonance curves were obtained by measuring and plotting vibration amplitudes at frequencies in the vicinity of the resonant frequency. Because damping values calculated from the resonance curves were unexpectedly high, the amplitude and frequency measurements were made by three different methods: (1) amplitude and frequency were obtained by analysis of oscillograph records; (2) strain-gage voltage output was measured by a highly damped meter; and (3) strain-gage signals were directed into a wave analyzer, which permitted determination of only the component corresponding to the first bending-mode frequencies of the blades. In the second and third methods, the frequencies were determined by observing the Lissajous figures produced on an oscilloscope by the combination of gage and oscillator signals.

RESULTS AND DISCUSSION

Inlet Disturbances

In most cases when blocks were placed in the inlet, vibration amplitude increased as indicated by the data points of figure 2. The highest stress thus produced, ±36,800 pounds per square inch, was a third-order vibration occurring in the third stage caused by three blocked inlet passages. This vibration was within the calculated range required for failure (reference 2) and possibly would have caused the blade to fail if the condition were permitted to continue for a considerable length of time. The effect of variation in number of blocked inlet passages on several orders of vibration of a first-stage blade vibrating in both first and second bending modes, shown in figure 3, indicates that when the order of vibration is divisible by the number of blocks, there is an appreciable increase in stress. The vibratory amplitude produced when the order of vibration is not divisible by the number of blocks is in most cases only slightly higher than those produced with unblocked inlets regardless of stage (fig. 4) or order of vibration (figs. 3 and 5).

The greatest effect of blocking was observed in the entrance stages directly behind the blocked inlet passages, as would be expected; appreciable effects, however, were present throughout the compressor even as far as the tenth stage (fig. 6). The actual magnitude of vibratory stress is meaningless for comparison between various stages because the speed at which the vibration occurs has such a pronounced effect on the amplitude. The data are therefore plotted as the percentage increase in stress as the result of inlet blocking over the stress observed without blocks. Each bar in figure 6 represents the average increase in vibratory stress for a given stage as determined from the stress measurements at several orders of vibration. The greatest effect of blocking occurred when the order of vibration was divisible by the number of blocks, and the maximum effect would be realized when the order of vibration was equal to the number of blocks.

The critical-speed diagram (fig. 7) shows the vibrations observed during normal operation without blocks. The fourth-order vibrations in the fifth stage selected for determining the effect of two diametrically placed blocks (fig. 1) successively positioned in every other inlet passage are plotted in figure 8(a). Averaging values for symmetrical positions with regard to the highest and lowest parts of the curve in figure 8(a) produced the smooth curve in figure 8(b). The spread of ±5 percent in stress level at various positions is accountable by slight differences in the individual inlet-air passages and the normal experimental accuracy of complicated strain-gage installations. When the inlet blocks are placed directly in the wakes of the front bearing supports, the maximum effect on vibrations should be produced. The peaks of the curves in figure 8(b), the rectilinear coordinates, and in figure 8(c), the polar coordinates, occur when the blocks are 16° counterclockwise of the bearing supports and not when directly behind them. Also the greatest reduction in amplitude of the fourth-order vibration should occur when the inlet blocks are halfway between the wakes of the four front bearing supports thus producing a strong eighth-order exciting force. The minimum amplitude also occurs when the blocks are displaced 16° from the midposition of the supports. A whirl velocity component must therefore exist in the air stream between the bearing supports (located at the 90° positions in fig. 8(c)) and the inlet guide vanes, moving the wake in the direction opposite to rotation of the compressor. The more significant characteristic evidenced from figure 8 is that certain vibrations in the compressor were suppressed by properly locating the blocks in the inlet of an axial-flow compressor. Approximate 10-percent increases or decreases in amplitude were created in the fifth-stage blades vibrating at the fourth-order critical speed, 14,250 rpm, by selectively placing two diametrically placed blocks in the engine inlet. Four blocks would have resulted in greater amplification or suppression of the fourth-order vibration.

Damping Characteristics

In order to realize compressor blades having acceptable vibration characteristics, knowledge of the damping characteristics of blades during normal engine operation is essential. No reliable experimental data on the subject exist, however, because of the difficulties of obtaining damping measurements unaffected by extraneous factors and of accurately analyzing the measurements obtained. Resonance curves of nonrotating specimens have frequently been analyzed by use of the following equations, which are only approximate:

$$\delta_{0.5} = \frac{\pi}{\sqrt{3}} \frac{\Delta f}{f_c}$$

$$\delta_{0.707} = \pi \frac{\Delta f}{f_c}$$

1154

where

δ log decrement of damping

Δf frequency spread at 0.5- or 0.707-peak amplitude, cycles per second

f_c frequency at peak amplitude of resonance curve, cycles per second

For lack of better methods, these equations were used to analyze resonance curves obtained from strain-gage measurements on blades of several stages. A representative curve for a fifth-stage blade is shown in figure 9. Analysis of figure 9 gave values of $\delta_{0.5} = 0.081$ and $\delta_{0.707} = 0.085$. Values of δ for blades of other stages were also approximately 0.08. Damping values obtained for stationary blades by the analysis of die-away curves (reference 2) averaged 0.015. The damping at standstill would constitute both root damping and internal damping of the blade material. Root damping would probably be reduced during rotation of the blade. It was assumed, however, that root damping remains unchanged and that the amounts of damping measured under static conditions by the analysis of die-away curves and under conditions of rotation by analysis of resonance curves can be compared. The difference between the damping of a stationary blade and that of a rotating blade was then used to represent the amount of aerodynamic damping present during normal engine operation.

Although the exact values of damping given are contestable, the relative magnitude should be reliable. The three methods of obtaining resonance curves gave satisfactory correlation and damping values from different blades at various speeds and orders also agreed favorably with each other. The conclusion that approximately four-fifths of the total blade damping present during operation of the unit investigated can be attributed to aerodynamic damping therefore seems reasonable.

Because aerodynamic damping is such a large part of the total damping capacity of a compressor blade subjected to normal operation, even doubling or tripling the material damping or the root damping, which in itself would be an outstanding accomplishment, would only slightly improve the over-all vibration characteristics of conventional axial-flow compressor blades. Aerodynamic damping is, however, unreliable as a safeguard during abnormal conditions such as stalling, blade flutter, or surging.

SUMMARY OF RESULTS

From the analysis of oscillograph records and measurements of the output from resistance-wire strain gages mounted on the blades of a 10-stage experimental axial-flow compressor, the following results were obtained:

1. Blocked compressor inlet passages altered the vibration characteristics of blades throughout the compressor.

2. Inlet obstructions could be so introduced as to decrease some blade vibrations.

3. Approximately four-fifths of the total damping capacity of the compressor blades during normal compressor operation was caused by aerodynamic damping.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

1154

REFERENCES

1. Meyer, André J., Jr., and Calvert, Howard F.: Vibration Survey of Blades in 10-Stage Axial-Flow Compressor. II - Dynamic Investigation. NACA RM E8J22a, 1949.

2. Meyer, André J., Jr., and Calvert, Howard F.: Vibration Survey of Blades in 10-Stage Axial-Flow Compressor. III - Preliminary Engine Investigation. NACA RM E8J22b, 1949.

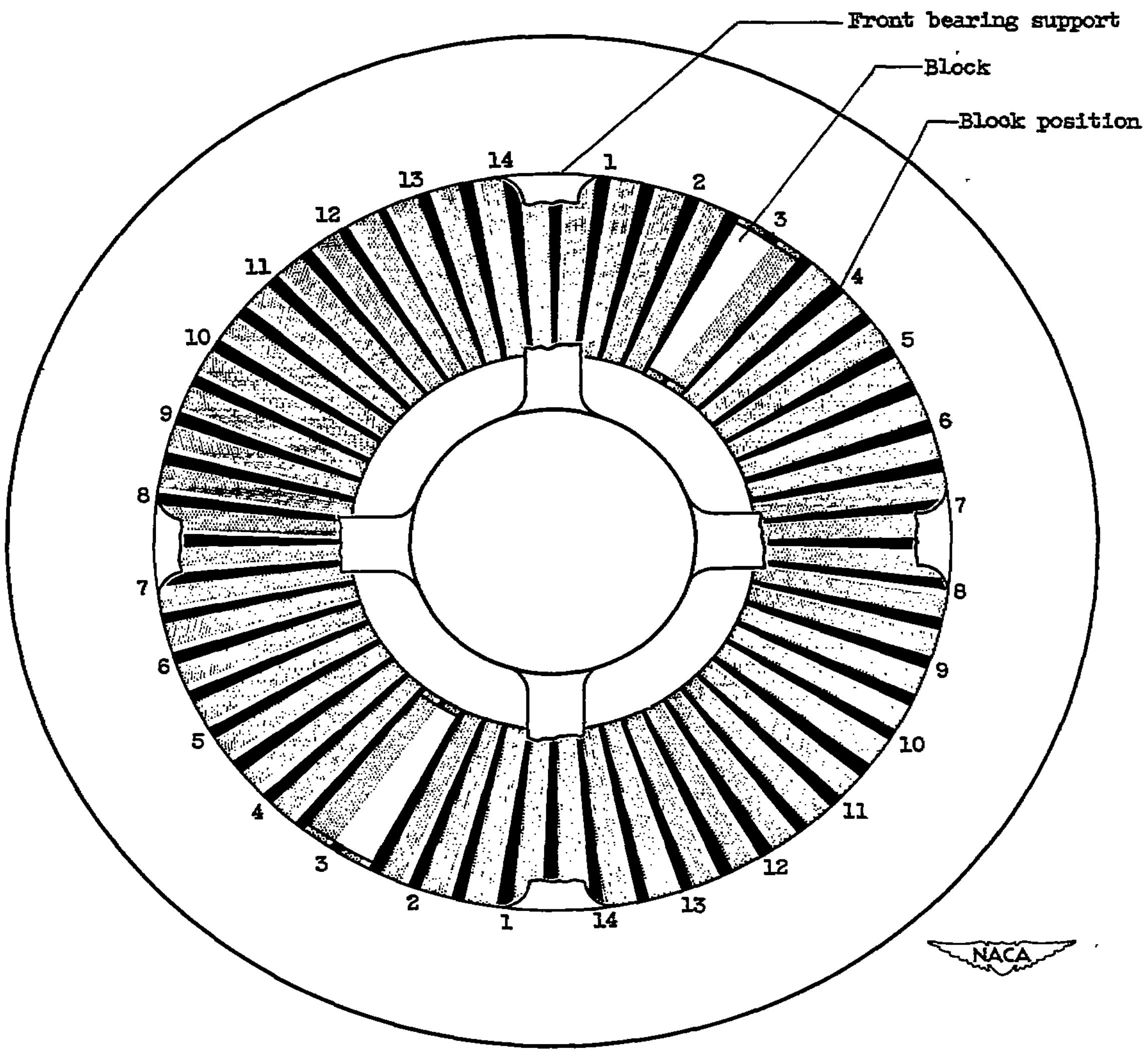


Figure 1. - Inlet of experimental compressor showing positions of blocked passages.

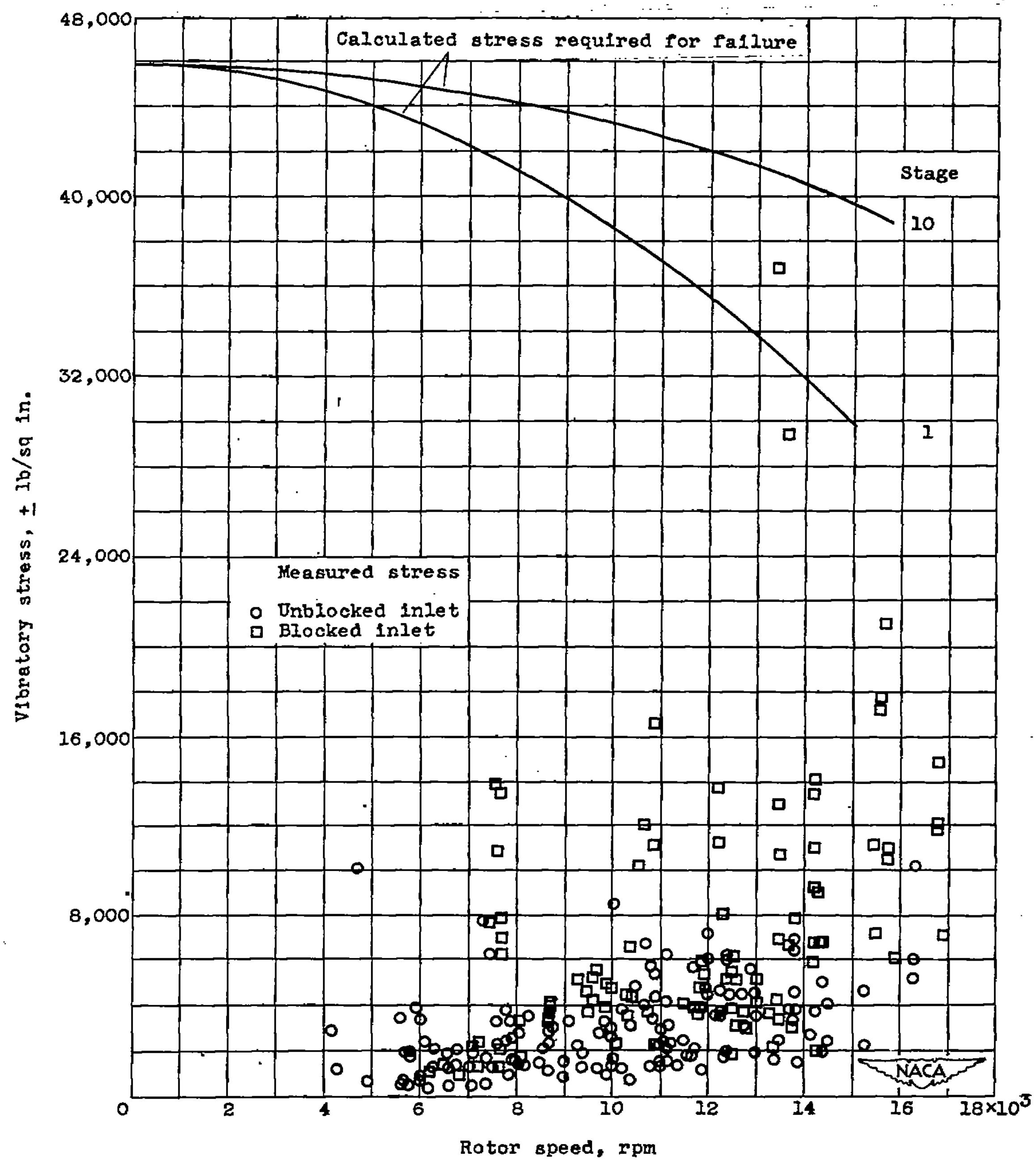
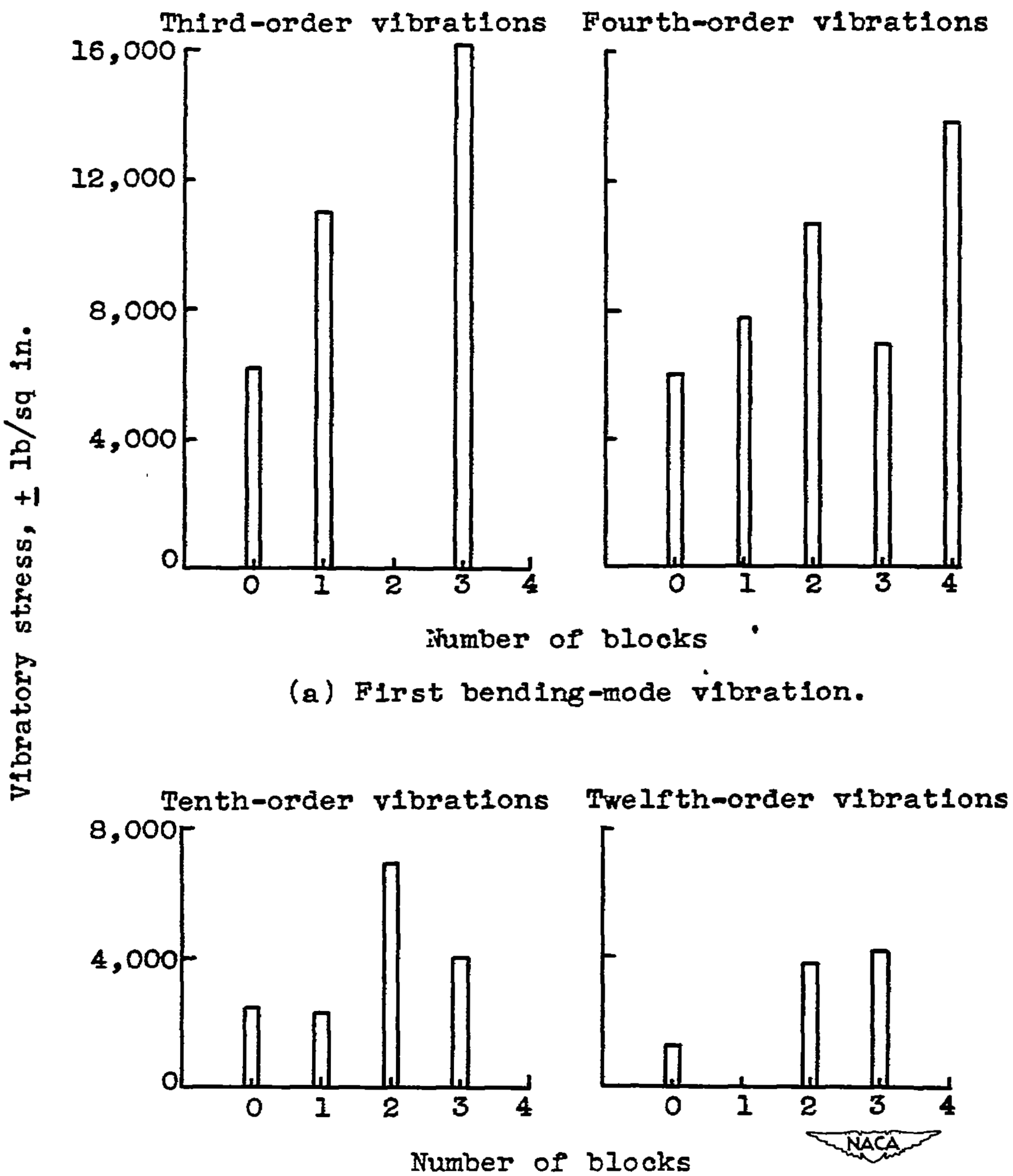


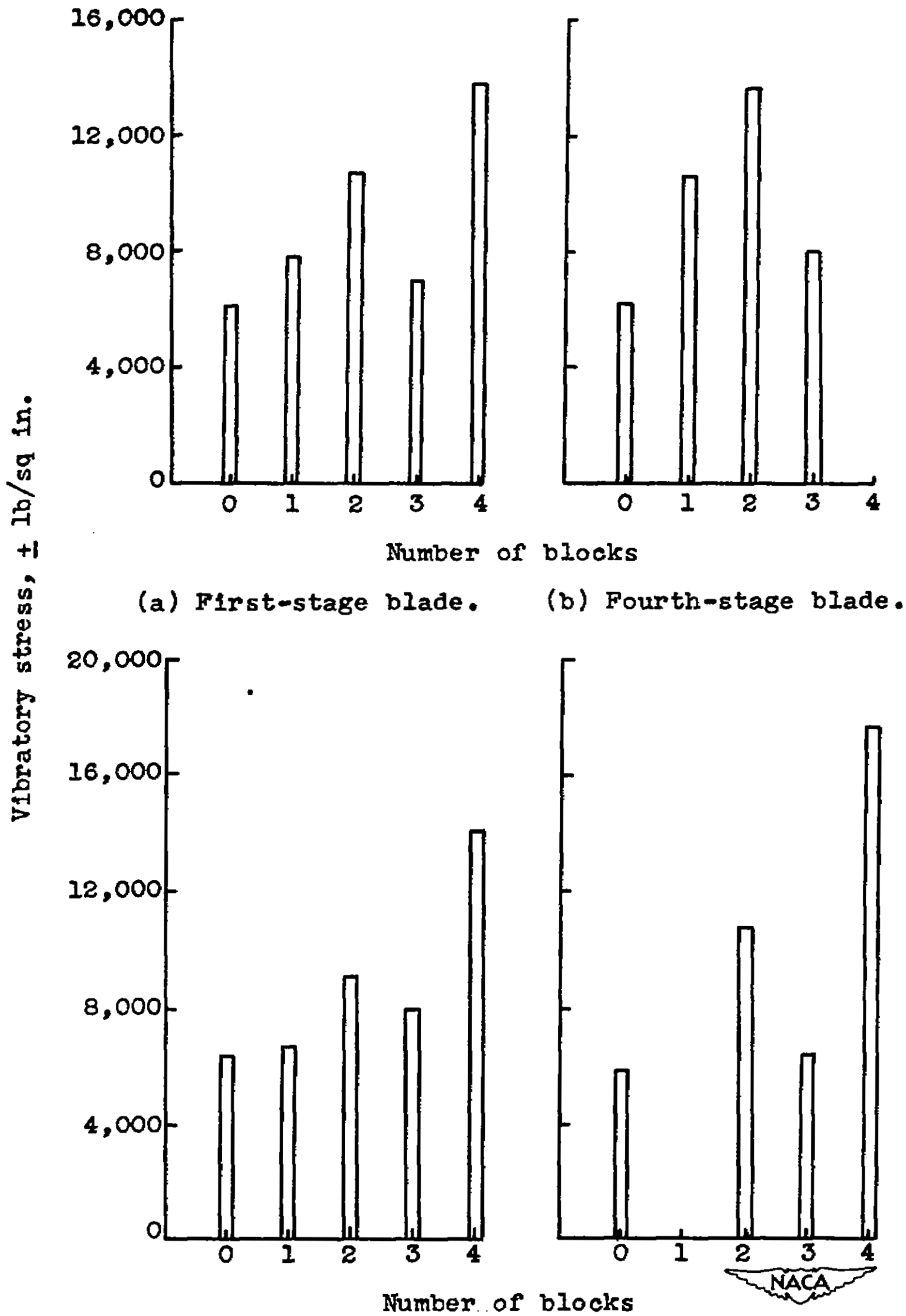
Figure 2. - Comparison of calculated vibratory stress required to produce failure and measured vibratory stress.



(a) First bending-mode vibration.

(b) Second bending-mode vibration.

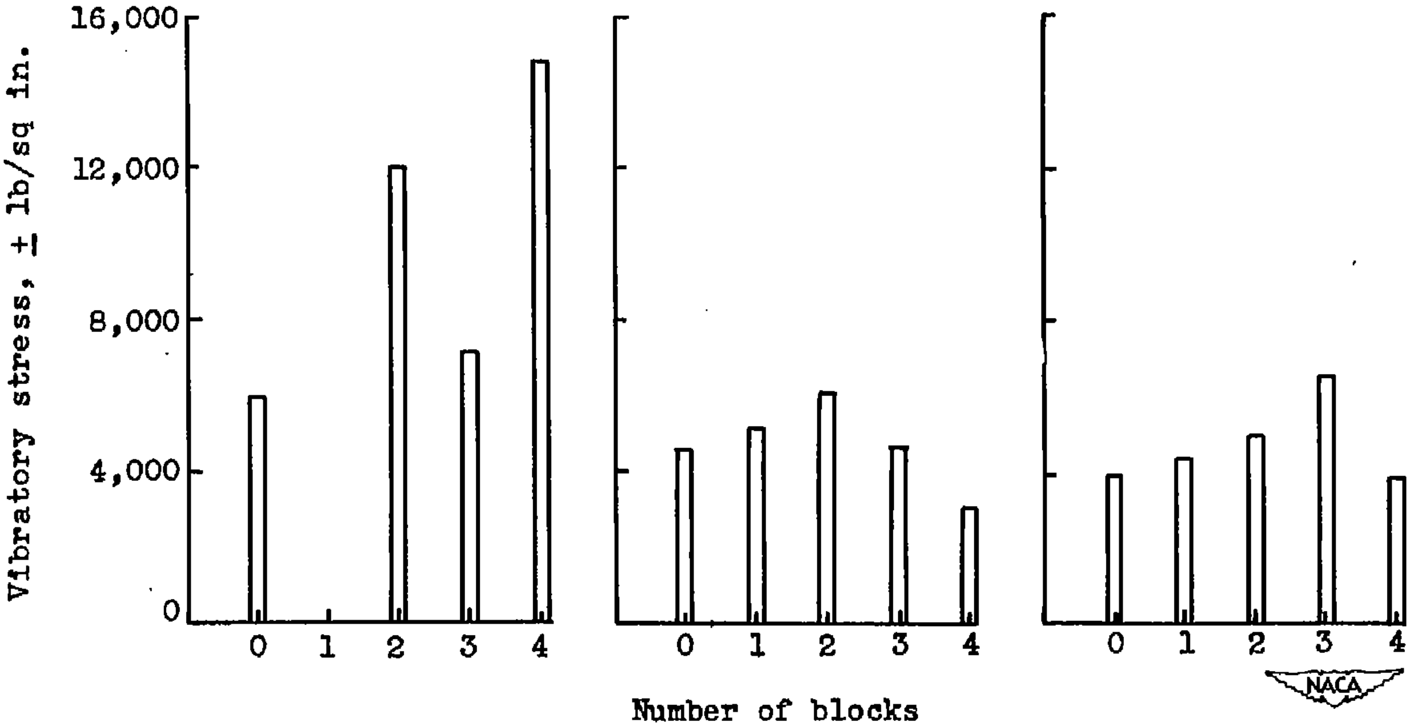
Figure 3. - Effect of number of blocked inlets on various orders of vibration in first-stage blade.



(a) First-stage blade. (b) Fourth-stage blade.

(c) Fifth-stage blade. (d) Seventh-stage blade.

Figure 4. - Effect of number of blocked inlets on fourth-order vibrations.



(a) Fourth-order vibrations.

(b) Fifth-order vibrations.

(c) Sixth-order vibrations.

Figure 5. - Effect of number of blocked inlets on various orders of vibration in seventh-stage blade.

1154

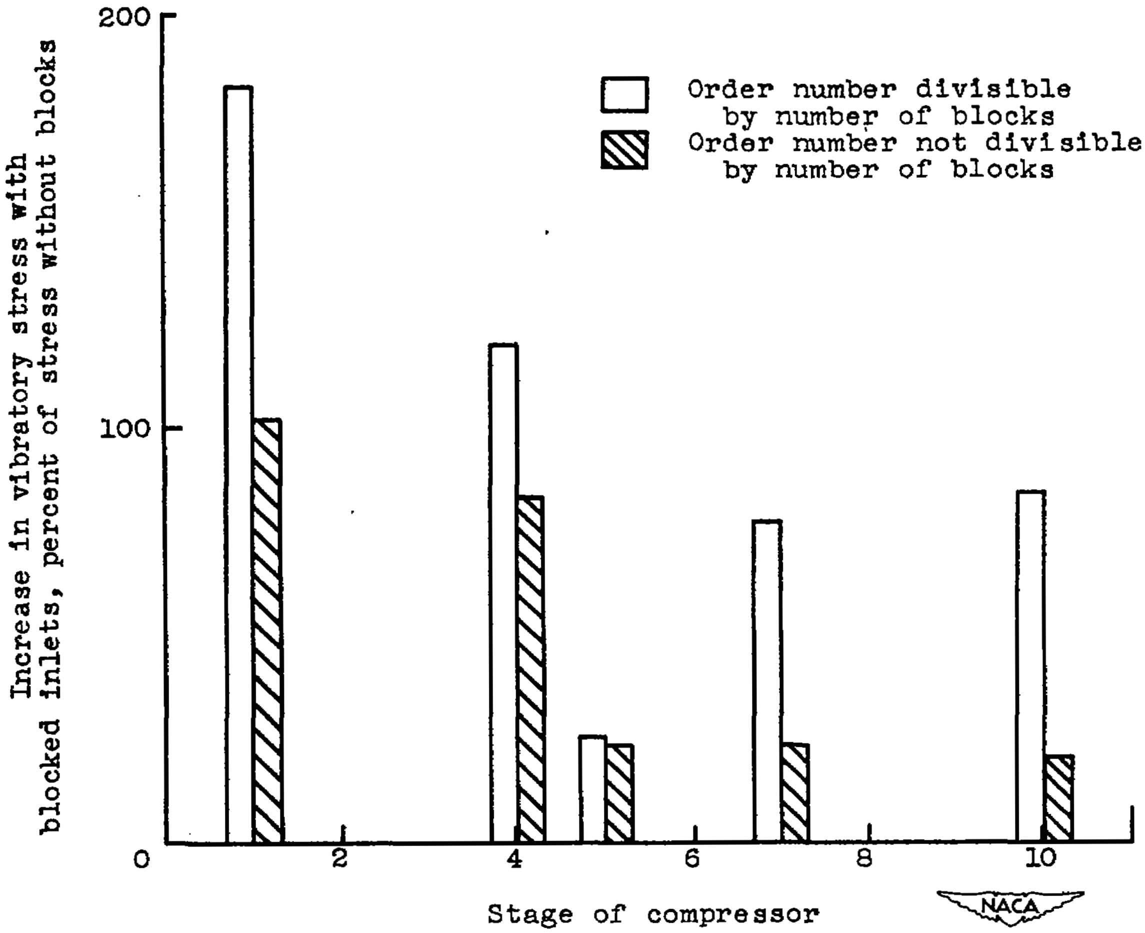


Figure 6. - Effect of blocked inlets on vibratory stress throughout compressor.

1154

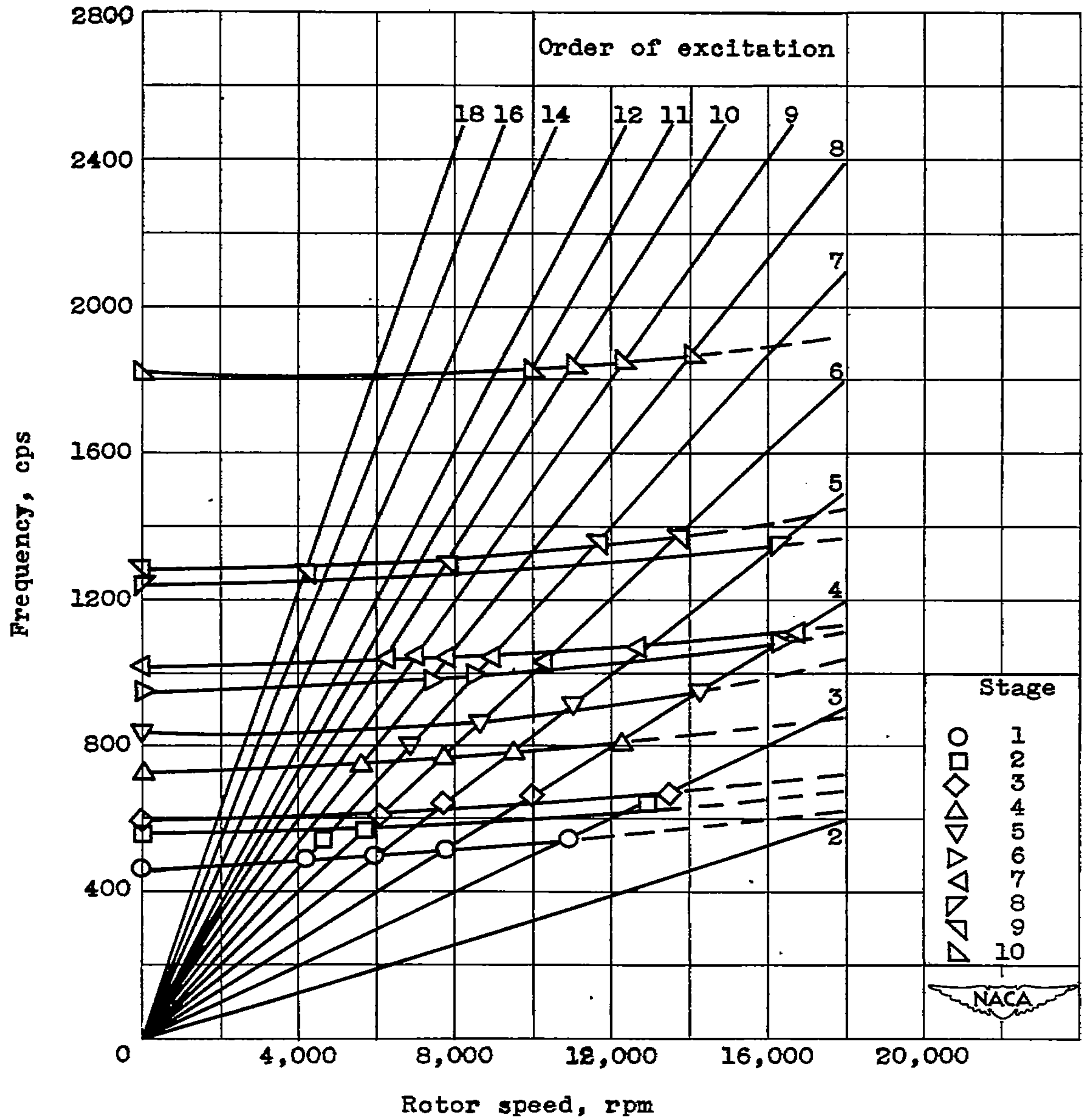
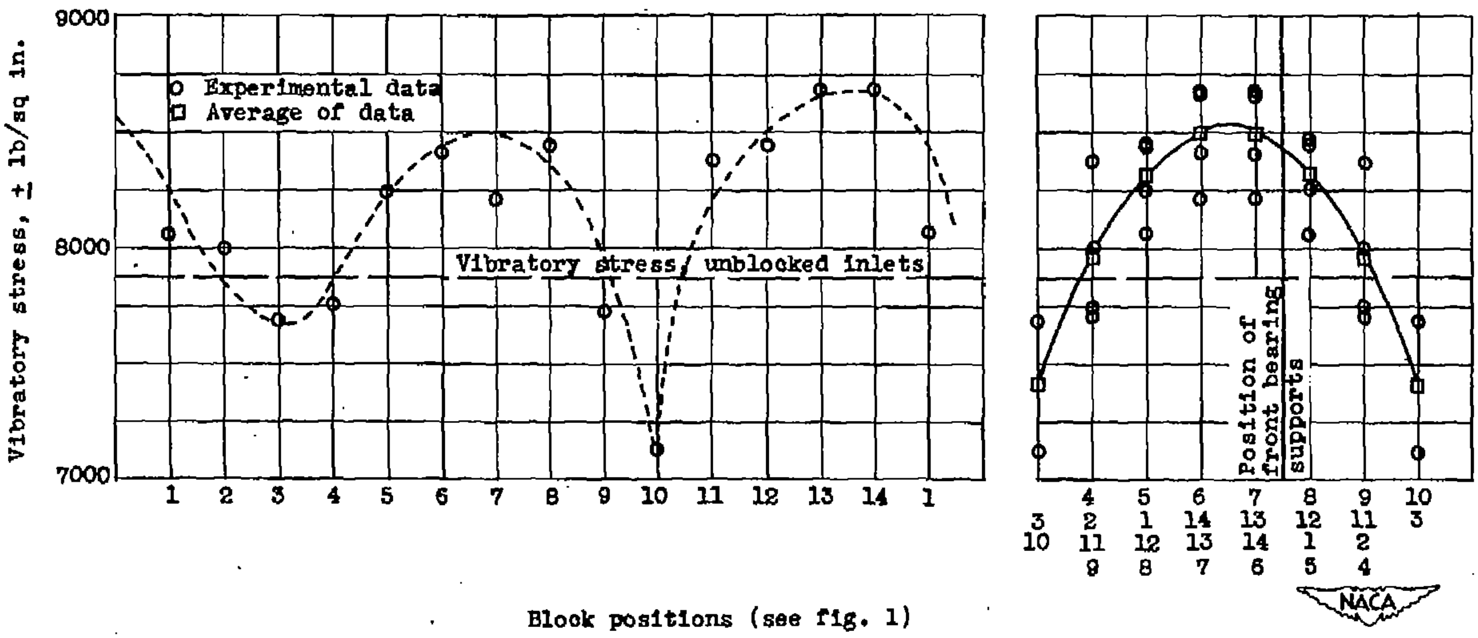


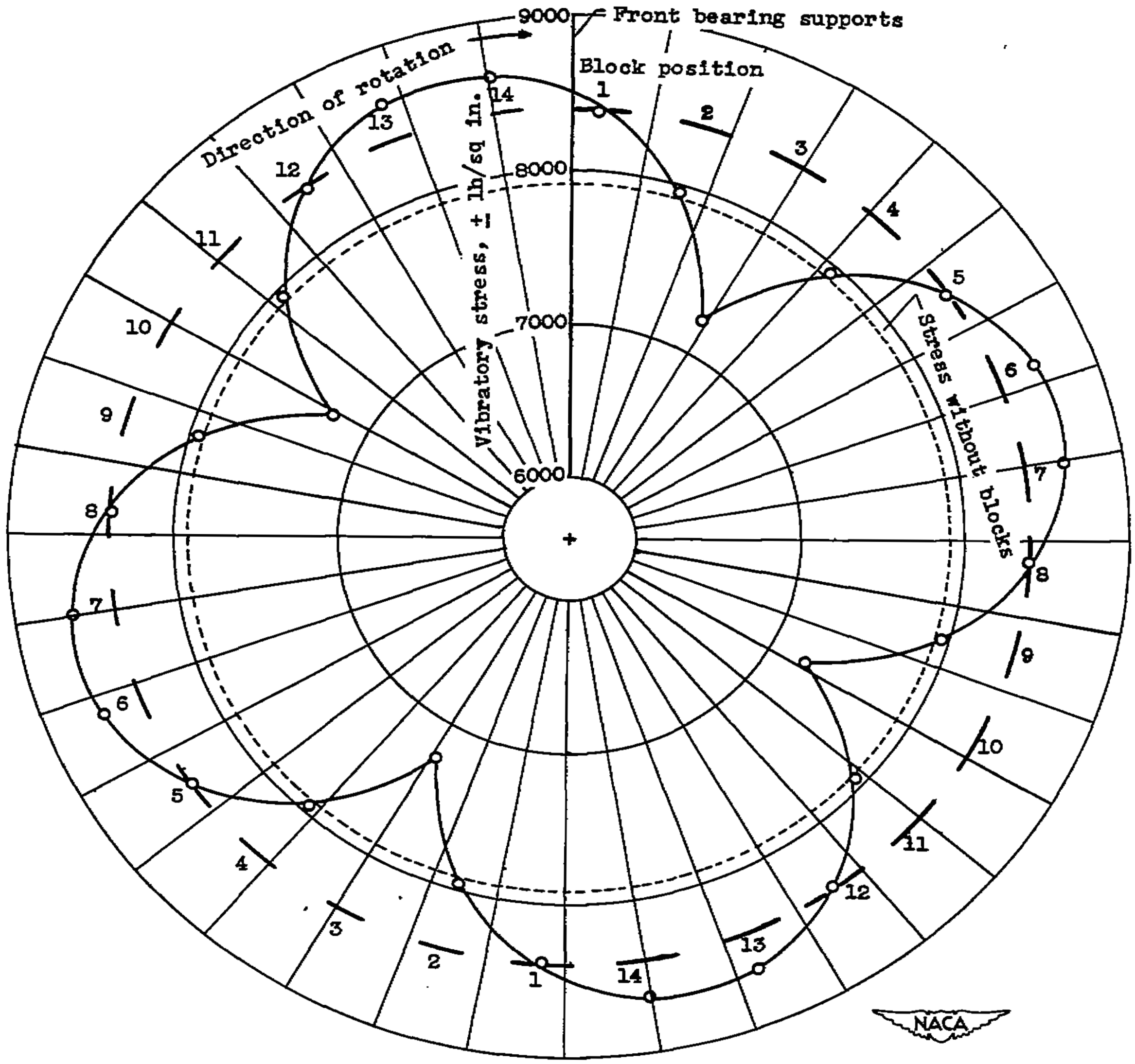
Figure 7. - Critical-speed diagram showing measured first bending-mode frequencies for all stages.



(a) Experimental data points.

(b) Averaged data points on rectilinear coordinates.

Figure 8 . - Effect of blocked inlet location on fourth-order vibrations in fifth-stage blade.



(c) Averaged data points on polar coordinates.

Figure 8. - Concluded. Effect of blocked inlet location on fourth-order vibrations in fifth-stage blade.

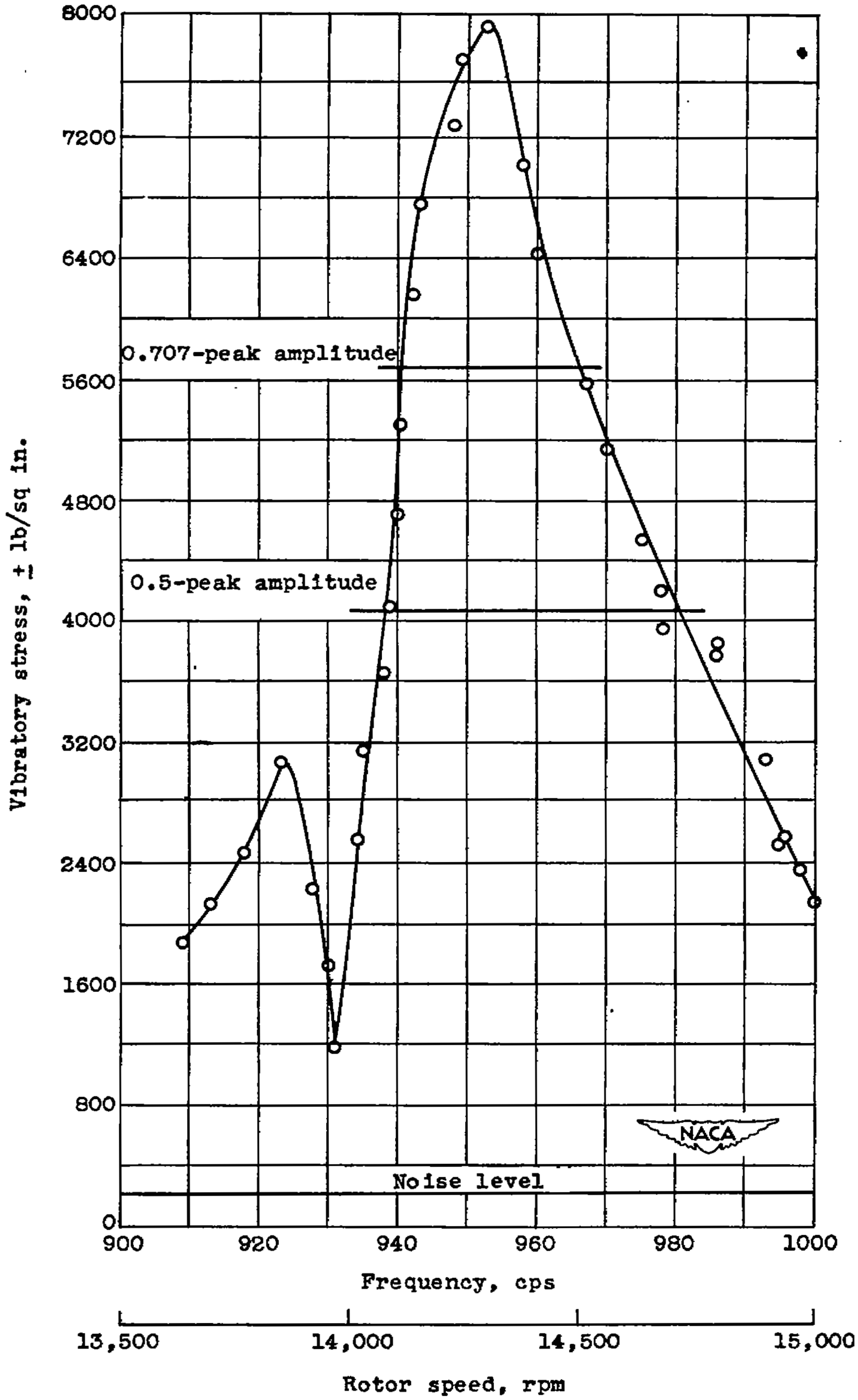


Figure 9 . - Resonance curve of fifth-stage blade.